

Comparison of the externally heated air valve engine and the helium Stirling engine



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ABSTRACT

A two-stroke, externally heated valve engine (EHVE) with a heater, a cooler and two blowers is simulated. The engine is entirely different from a typical Stirling engine. The pressure ratio p_{max}/p_{min} of its cycle is higher, but the engine volume and the mean value of the heat exchanger wall temperatures are the same. The power and efficiency of the EHVE and Stirling engines under the same maximum pressures are compared. The results show that the EHVE engine reaches almost the same level of performance as the Stirling engine, while using only available atmospheric air, rather than helium.

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1. Introduction

In this paper, a new type of externally heated valve engine (EHVE), fully described in [1], is examined and quantitative results of its performance are presented. Schematic views of the engine are shown in Figs. 1 and 2.

Stirling engines are well known externally heated devices and remain a subject of current research [2–6]. Such engines have similarities to the EHVE, including: external heating; high thermal efficiency; multi-fuel capability; low noise generation; and low pollution when a combustion process is used as the heat source. A general feature of the EHVE and Stirling engines is the ability to use a range of heat sources. When a combustion process is used for this purpose, the burning arrangement may be optimised for a particular fuel. However, heat sources are not limited to combustion processes only, but can involve any type of energy capable of reaching the required temperature.

There are many papers showing experimental results of different Stirling (or similar externally heated) engines designed to use various alternative sources of energy, see [7–24], although not all were intended to reach high performance levels. Applications range from demonstration devices such as [23], through general purpose machines [7,21], solar heated [13], marine use [17,26], to potential outer space energy conversion [25]. The reader can find a comprehensive review of such solutions in [26–29] and a

summary of the performance of a number of existing machines in [4].

Numerical simulations of the EHVE operation are compared with the experimental results presented for the Stirling engine described in [2,3] (1987–1988). The engine has a compact mechanical form and comparatively high power and efficiency at a total volume of 588 cm³ enclosed in four double-action cylinders with four piston rods. The working gas is helium. Later Stirling engines described in [4–6] (2001–2003) have analogous or even lower effectiveness at similar design parameters. An initial comparison of the EHVE and an example of the Stirling engine has been made already in [30], but taking into account the later results developed in [31], such evaluation could be too optimistic for the EHVE. That evaluation is expanded and completed here.

2. Design aspects of the EHVE

The EHVE in the earlier form as shown in [30] exists as a prototype version ([31], Fig. 3). The experimental results have already proven the concept. In the prototype, an electric heating device was applied. However, independently of its type, it resulted in insufficient heat being delivered during the period when the engine valves were closed. An intensive flow of air is required in the heater throughout the complete cycle. Therefore, two centrifugal blowers B_1 and B_2 were added to the model of the EHVE in order to improve the heat exchange rate. Their power requirements are minimal but raise the heat exchange level when the engine valves are closed. The version of the EHVE discussed here is equipped with a single heater and a single cooler. The volumes

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Nomenclature

A	surface area (m^2)	η	efficiency
D, d	diameters (m)	κ	isentropic expansion index
h_{Sbi}	enthalpy isentropic increase of blowers (J/kg)	ω	angular velocity of the crankshaft (rad/s)
L	length of tubes (m)	Subscripts B_1, B_2	blowers
$\dot{m}_j$	mass flow rate (kg/s)	C	compressor
p	pressure (N/m^2 , bar)	Cl	cooler
P	power (W, kW)	Cl_2, Cl_4	collectors of the cooler
n	number of tubes	E	expander
Q	heat (J)	f	friction
$\dot{Q}_A$	heat flow in the surface area (W)	H	heater
R	specific gas constant (J/kg K)	H_1, H_3	collector of the heater
Re	Reynolds number	i	number of the blowers ($=1, 2$)
S	entropy (J/kg K)	j	number of the valve ($=1, 2, 3, 4$)
T	temperature (K)	k	opening ($k=0$), closing ($k=c$)
V	volume (m^3)	max	maximum
α	crankshaft angle (rad, $^\circ$)	PH	pre-heater
$\alpha_{j,k}$	angle of valves (rad)	w	wall

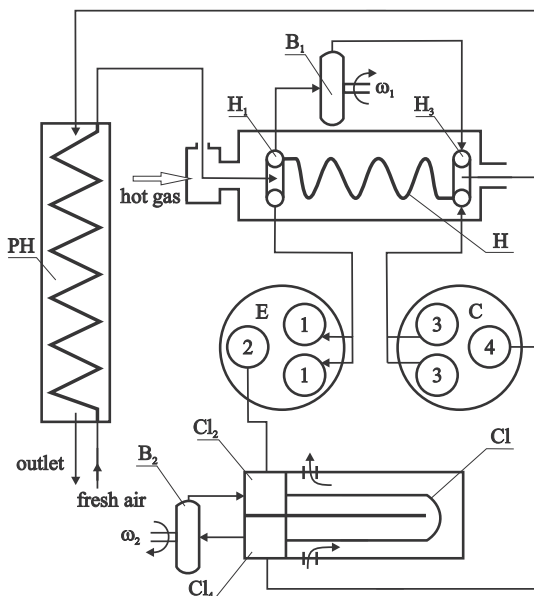


Fig. 1. Scheme of the transversal cross-section of the engine E – expander, C – compressor, H – heater, Cl – cooler, B₁, B₂ – blowers, 1, 2, 3, 4 – valves, H₁, H₃ – heater collectors, Cl₂, Cl₄ – cooler collectors, PH – pre-heater.

of each heat exchanger in the EHVE are at least 3–5 times larger than the volumes of the cylinders. The inlet and outlet collectors of the heater and the cooler are shown in Fig. 1. Their volumes may play a role of settling chambers for the working gas flow (equalizing pressure and temperature values in the exchangers in time).

An important advantage of this configuration is use of the most easily available working gas—atmospheric air. Also, it is possible to utilise any source of heat able to deliver an appropriate temperature to the heater, that is, from solar energy, through combustion of other media, to nuclear power. This makes the EHVE a suitable solution when available resources of oil and gas run out. The high temperature of the heating gas leaving the EHVE allows for the installation of an additional heat exchanger, a pre-heater (PH), improving thus efficiency, see Fig. 1 and Section 3. The earlier EHVE scheme, shown in [1], did not include such a device.

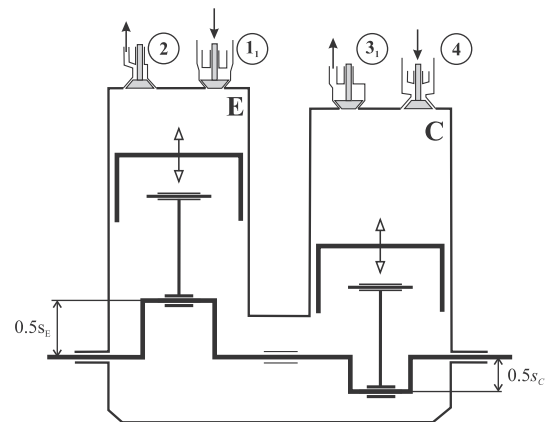


Fig. 2. Scheme of the longitudinal cross-section of the engine.

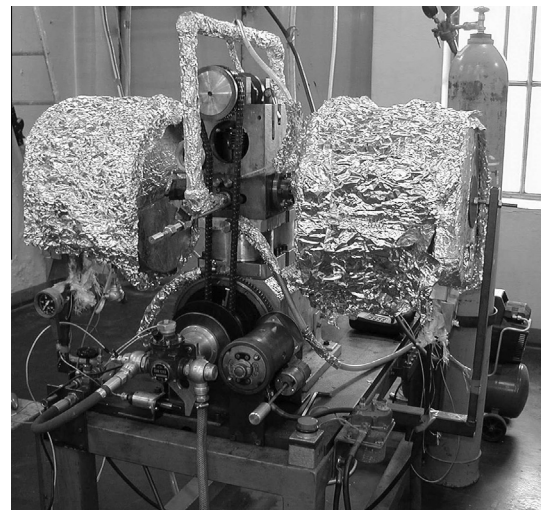


Fig. 3. General view of the EHVE prototype. Engine block in the middle, heat exchangers at sides – two electric heaters have been used in the experiments performed in 2001, see [4].

The EHVE used in the comparison is a two-stroke type device. It is equipped with two cylinders, each of which plays a different role (Figs. 1 and 2). The expander E and the compressor C are situated between two external heat exchangers, the heater H and the cooler Cl, with the phase angle between them equal to 180° . Both of the heat exchangers, H and Cl, are additionally equipped with their own blowers B_1 and B_2 raising the rate of heat exchange. The gas flow inside the exchangers is assumed to be turbulent even when the engine valves are closed. The engine works in a closed cycle. Its valves are operated by means of a typical camshaft. Additionally, the EHVE features a conventional crankshaft and a conventional oil lubrication system, which is an advantage for manufacturing. As in the internal combustion (IC) and existing Stirling devices, one can expect an increase of performance in a multi-cylinder arrangement of the engine [9].

The pressure ratio $p_{\max}/p_{\min}$ of the EHVE and the Stirling engines considered is different. For the EHVE, it is greater than 5, while for both of the Stirling engines considered, it is less than 2. For this reason, comparisons cannot be performed with equivalent minimum pressures. For the Stirling engines, the minimum pressures are equal to 60, 82 and 110 bar, but in the EHVE such minimum pressures would imply unrealistically high values of the respective maximum pressures, which may be not acceptable from an engineering point of view. The new form of the EHVE with blowers is therefore compared with the Stirling engines using the same maximum working pressure.

3. Simulations of the EHVE

The model of the EHVE and its mathematical realisation is described in detail in [1]. We repeat only the important elements here. These are time-dependent relations for governing the engine action, in which dependence on spatial variables has been omitted, so the model is described and simulated in the time domain only. In both heat exchangers, the spatial changes are important because, e.g., the temperature of the working air inside the exchangers in fact varies on the way from the inlet to outlet collectors. A simpler approach to the problem is also possible and it has been applied in [1].

The heat exchangers can be described as follows. The heater and the cooler are counter-flow heat exchangers, where the temperature gradient remains nearly constant along their length. The heat exchangers are treated as time-dependent only, assuming that all important changes are the same along the overall length of the exchangers and considering each of them in a single plane only. The plane is situated at the outlet of the exchanger tubes. The approach used here assumes the application of maximum wall temperatures of the heater tubes and minimum temperatures of the cooler wall tubes. This method of analysis of heat exchangers is sufficient to demonstrate the advantages resulting from use of such blowers. Such a treatment of the parameters is an approximation, but it allows the use of a moderate number of equations in the time-dependent model.

The blower in the loop of hot and cool air circulation ensures that the Reynolds number calculated for the exchanger tubes is not less than 10^4 , when both valves 1 and 3 remain closed.

The work of the expander and the compressor is calculated for an angle $0^\circ \leq \alpha \leq 360^\circ$, in the same way as for the 2-stroke engine:

$$L_E = \int_{\alpha=0^\circ}^{\alpha=360^\circ} p_E(\alpha) \frac{dV_E}{d\alpha} d\alpha, \quad (1)$$

$$L_C = \int_{\alpha=0^\circ}^{\alpha=360^\circ} p_C(\alpha) \frac{dV_C}{d\alpha} d\alpha. \quad (2)$$

The pressures in the above formulas are determined according to [1], taking into account losses in the valves.

The work of the engine is the sum of:

$$L = L_E + L_C - L_{B1} - L_{B2} - L_f - L_{\text{sup}}, \quad (3)$$

where the values of work of the blowers are ($i = 1, 2$):

$$L_{Bi} = \frac{2\pi \dot{m}_{Bi} h_{SBi}}{\omega_{Bi} \eta_{Bi}}. \quad (4)$$

L_f is the work due to friction in the machine, and, L_{sup} is the work lost through all additional devices attached to the EHVE. Frictional losses in modern IC engines are observed to be between 3% and 8% of total horsepower. As the EHVE can use almost the same oil lubrication system as an IC engine, similar frictional loss values were applied. Its state-of-the-art design can reduce amount of the losses considerably.

The symbols used are explained in the nomenclature, and an enthalpy isentropic increase of blowers is calculated from:

$$h_{SBi} = \frac{\kappa}{\kappa - 1} RT_{bBi} \left(\Pi_{Bi}^{\frac{\kappa-1}{\kappa}} - 1 \right). \quad (5)$$

A complete power output of the externally heated 2-stroke engine is determined as:

$$P = \frac{\omega}{2\pi} L. \quad (6)$$

The heat delivered inside the heater (following the theoretical model described in [1]) is:

$$Q_H = \int_{\alpha=0^\circ}^{\alpha=360^\circ} \dot{Q}_{AH} \frac{d\alpha}{\omega}. \quad (7)$$

The efficiencies of the engine may be defined as:

$$\eta = \frac{L}{Q_H}, \quad \eta_{PH} = \frac{L}{Q_{H-PH}}. \quad (8)$$

The calculation of the value of Q_H is made without an additional heat exchanger. However, the temperature of the gas heating the heater tubes at its outlet (Fig. 1) would be very high, so a pre-heater may be applied as described in [2,3]. The pre-heater being an additional heat exchanger raises the inlet temperature of the heating gas and causes considerable savings in heat requirements (decreasing the denominator value in Eq. (8) from Q_H to Q_{H-PH}). The efficiency of the EHVE should increase significantly in this case.

4. Comparison of the air EHVE to helium Stirling engines

The numerical simulation of the EHVE then requires a solution of 19 non-linear ODE's with appropriate switching conditions, supplemented by a set of nonlinear algebraic equations describing the behaviour of elements of the model. Stationary, periodic solutions to the engine operation at a given, constant angular velocity of the engine are presented. The simulation program uses a C++ language library of procedures and solvers (Runge–Kutta fourth-order method) [32]. All simulations started from identical, fixed initial conditions and continued until the solution reached the point where the integrated values differed by no more than 0.01% from those obtained at the end of the previous cycle. A good agreement of the integrated values with the experimental results for the prototype engine was found, as published earlier in [31]. Typically, a stable solution was reached after 150 cycles of simulated crankshaft revolutions.

The calculations were performed for the ratio of volumes $V_C/V_E = 0.75$, which was found to be the optimal value for the case when $V_H \approx 5V_E$, see [1]. The EHVE has the same expander volume V_E as the Stirling engine [2,3], which is a double-action system.

Other essential data, different to those presented in [1], are as follows:

$$\begin{aligned} d_E &= 0.0900 \text{ m}, & s_E &= 0.0925 \text{ m}, & h_{0E} &= 0.0025 \text{ m}, \\ d_C &= 0.0900 \text{ m}, & s_C &= 0.0694 \text{ m}, & h_{0C} &= 0.0017 \text{ m} \end{aligned}$$

– representing dimensions in the volumes of E and C.

Dimensions of the heat exchangers (d – pipe diameter, l – pipe length, n – number of pipes, and, V – the exchanger volume) have been assumed as:

$$\begin{aligned} d_H &= 0.01 \text{ m}, & l_H &= 2.00 \text{ m}, & n_H &= 15, & V_{H1} &= V_{H3} = 0.001 \text{ m}^3, \\ d_{Cl} &= 0.01 \text{ m}, & l_{Cl} &= 2.00 \text{ m}, & n_{Cl} &= 15, & V_{Cl2} &= V_{Cl4} = 0.001 \text{ m}^3. \end{aligned}$$

The value of Δp_{H3} depends on the design of the collectors and the heat exchangers. Here, all values of the Δp 's have been fixed at 10 kPa. The mass flow rates of the blowers have been estimated at $\dot{m}_{B1} = 0.1$ and 0.2 kg/s. Other important blower parameters are: input to output pressure ratio $\Pi_{B1} = 1.01$, and efficiency $\eta_{B1} = 0.7$.

The work of friction during the cycle is estimated as (derived and adjusted to the EHVE from [33–35]):

$$L_f = 40 + 0.8 \frac{\omega}{2\pi}.$$

The power for all auxiliary devices – L_{sup} , is supplied by an electric motor rotating at 3000 rpm, independently of the EHVE rotation, and is also taken into account. For this purpose, the calculation assumes a power consumption of 1000 W.

The temperature of the EHVE heater walls is assumed to increase with the engine rotational speed from 1100 to 1250 K and it is related to the amount of hot heating gas delivered, see Table 1. This temperature reflects the limited amount of data given in [2,3] for the Stirling engines. The minimum temperature of the cooler wall is constant and equal to $T_{wCl} = 298$ K based on standard parameters of the cooling water supply and data from [2,3]. The coefficients of heat conduction, dynamic viscosity and specific heats are taken from tables of physical properties of air.

The simulation program, based on the model of the EHVE described in [1] has been used for calculations for the parameters reflecting the data available for the two Stirling engines presented in [2,3] at selected rotational speeds from 500 to 2000 rpm.

In Fig. 4, a difference in the area enclosed by the p - V curves can be interpreted as the useful work provided by the EHVE. Fig. 5 illustrates a T - S diagram of the EHVE and a short description of the operation cycles. The cycle shown in Fig. 5 is a new type of the heat exchange cycle. The statement that it is similar to the Joule cycle, which was given in [1], should be treated as a simplification.

Use of a pre-heater does of course reduce the energy requirements of the heater. All details presented in [2], (earlier than [3]), and even the pre-heater described there are different when compared to [2]. Therefore, it has been assumed $Q_{H-PH} = 0.80Q_H$ for [3] and $Q_{H-PH} = 0.85Q_H$ for [2]. These assumptions are rather modest and can be easily met in real installations. The efficiency of the EHVE in this case is obviously higher than in the case without a pre-heater and is presented in Figs. 6–8. It is important that the Stirling engines against which the EHVE is compared make use of pre-heaters [2,3]. The heat delivered to the EHVE reflects the

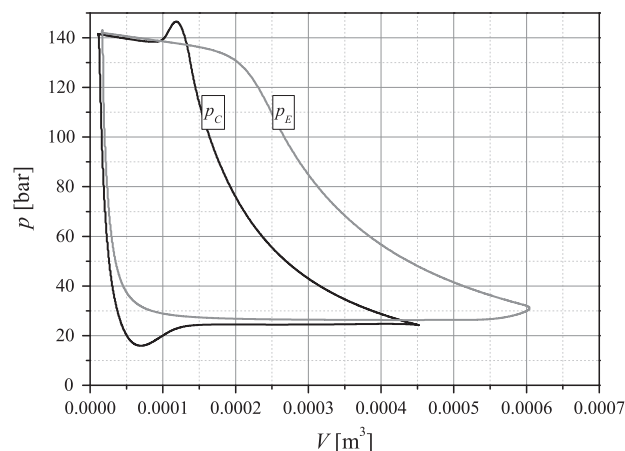


Fig. 4. p - V diagram for the EHVE at 1500 rpm, $p_{max} = 140$ bar and $\dot{m}_{B1} = 0.1$ kg/s. Data in Section 3.

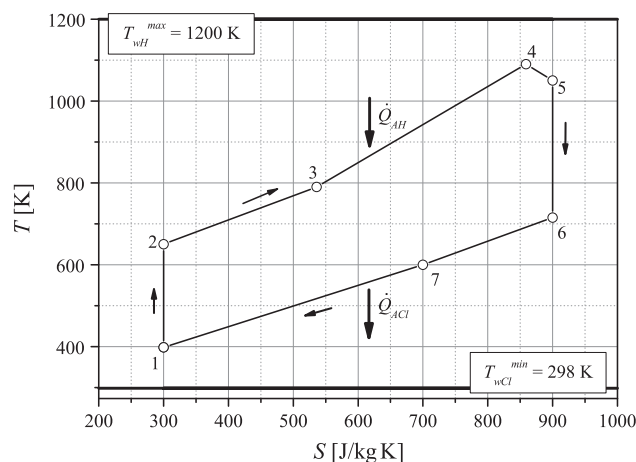


Fig. 5. Approximate T - S diagram for the EHVE at 1500 rpm, $p_{max} = 140$ bar and $\dot{m}_{B1} = 0.1$ kg/s. 1–2: isentropic compression inside the closed C volume, 2–3: compression by C, heating by H and mixing in the collector H_3 by the blower B_1 , 3–4: heating inside the H volume, 4–5: expansion in E with heating at opened valve 1 until its closing, 5–6: isentropic expansion inside the closed E volume, 6–7: flow to the collector Cl_2 , expansion and mixing with the collector Cl_2 by the blower B_2 , 7–1: cooling inside the Cl volume.

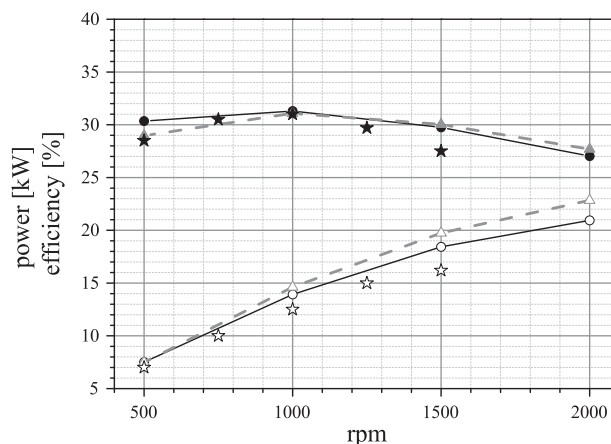


Fig. 6. Comparison of the EHVE and AISIN Stirling engines at the maximum working pressure $p_{max} = 102$ bar. Power values presented with empty markers and efficiency with the filled ones. Circles on solid lines represent the EHVE with its blowers working at $\dot{m}_{B1} = 0.1$ kg/s, top-triangles in grey, dashed line with blowers at $\dot{m}_{B1} = 0.2$ kg/s, whereas star symbols stand for the AISIN engine [3].

Table 1

Heater wall maximum temperatures of the EHVE as a function of rpm. The heater wall mean temperature is the same for both the air EHVE and helium Stirling engines [2,3] and equal to 1023 K.

n_{EHVE} (rpm)	500	1000	1500	2000
T_{wH} max (K)	1100	1150	1200	1250

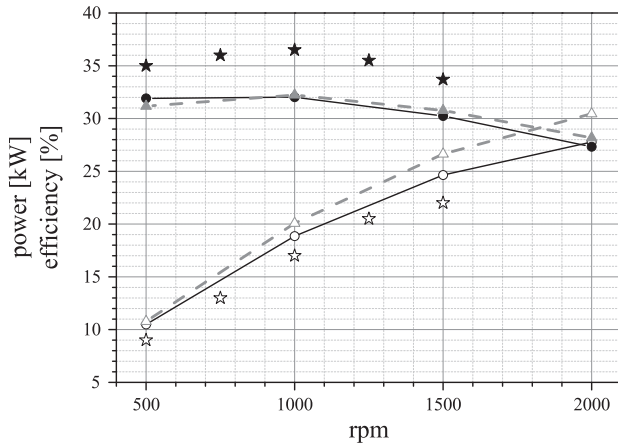


Fig. 7. Comparison of the EHVE and AISIN Stirling engines at the maximum working pressure $p_{max} = 140$ bar. Power values presented with empty markers and efficiency with the filled ones. Circles on solid lines represent the EHVE with its blowers working at $\dot{m}_{B1} = 0.1$ kg/s, top-triangles in grey, dashed line with blowers at $\dot{m}_{B1} = 0.2$ kg/s, whereas star symbols stand for the AISIN engine [2].

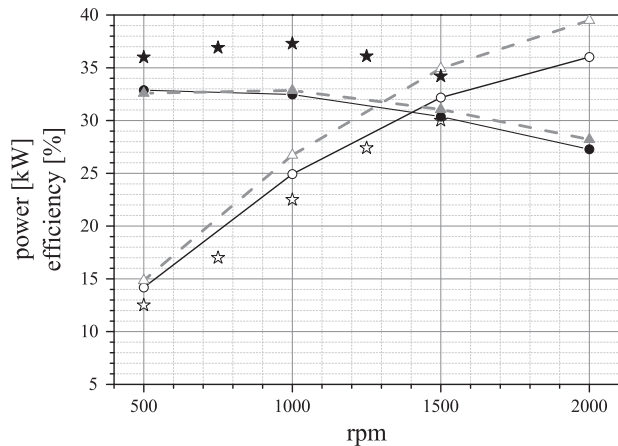


Fig. 8. Comparison of the EHVE and AISIN Stirling engines at the maximum working pressure $p_{max} = 190$ bar. Power values presented with empty markers and efficiency with the filled ones. Circles on solid lines represent the EHVE with its blowers working at $\dot{m}_{B1} = 0.1$ kg/s, top-triangles in grey, dashed line with blowers at $\dot{m}_{B1} = 0.2$ kg/s, whereas star symbols stand for the AISIN engine [2].

unimproved pre-heater and is then assumed to be equal to $Q_{H-PH} = 0.85Q_H$.

The comparisons presented below have been made between the numerical results obtained here and the experimental results referred to the helium Stirling engine described in [2,3]. The results are presented in Figs. 6–8 for the same engine volume, rotational frequency and maximum pressures. The maximum pressure values of 102, 140 and 190 bar for the Stirling engine cycle have been derived from the data given in [2,3]. Taking into account the results presented in [3], the heat exchangers, including a pre-heater are improved with efficiency values about 4% higher than those quoted in [2].

The blowers B_1 and B_2 working at a flow rate of $\dot{m}_{B1} = 0.1$ kg/s are necessary for efficient operation of the EHVE. A further increase in the blower flow rate is not practical, see (Figs. 6–8). The reason is an increase in the power consumed by both blowers. The comparison also shows that the EHVE for all values of p_{max} at $\dot{m}_{B1} = 0.1$ kg/s can produce slightly more power than the equivalent helium Stirling engine. It must be noted that this is achieved with air as a working gas rather than helium.

A comparison of the efficiency levels of both engines is more complicated. The maximum efficiency of the Stirling type shows a distinct peak at 1000 rpm, whereas the efficiency plot for the EHVE is rather flatter. In Fig. 6, where heat exchangers (basic type in the case of Stirling engines) are almost the same, it is seen the efficiency of the EHVE is nearly equal. In Figs. 7 and 8, where heat exchangers in the Stirling engine are improved, the EHVE efficiency is lower on average by about 4%.

The presence of valves in the EHVE also causes some disadvantages, which are as follows:

- (1) a short period of opening of valves 1 and 3, as in [1], $\Delta\alpha_1 = 90^\circ$, and $\Delta\alpha_3 = 80^\circ$.
- (2) the blower B_1 works at high temperature and pressure conditions (Fig. 1).

The period of the valve opening phase is a challenging but feasible engineering task. The necessary work related to valve timing is still in progress [36]. The heater, not discussed here in detail, is also a demanding component of the EHVE. High working pressure and temperature are the critical factors in its design. Continuous improvements of heaters used in the Stirling engines however allow us to anticipate their availability for the EHVE without any major changes. Some examples of heaters and heat exchange problems are presented in [37–40].

5. Conclusions

A two-stroke externally heated valve engine called the EHVE, entirely different from the well known Stirling type, has been investigated by numerical simulations. Two counter-flow heat exchangers, a heater and a cooler, and a pre-heater are employed in the configuration. These have been supplemented by two blowers B_1 and B_2 in order to increase the rate of heat exchange.

The results have been compared with the experimental data obtained for very good examples of helium Stirling engines [2,3] for similar maximum pressures of their respective cycles. The same engine capacities, heater wall temperatures and working frequencies have been assumed, see Figs. 6–8. The differences in pressure ratios p_{max}/p_{min} of the compared engines being about 5:2 are mainly caused by the presence of valves and design of dead space. The Stirling engine has no valves and large dead space, contrary to the configuration of the EHVE.

The EHVE presents an improvement in the area of externally heated engines, especially over the Stirling designs in terms of power, losing only some efficiency under similar working conditions. The most important advantage of the new design is such that these results have been achieved using atmospheric air rather than helium. We estimate that using helium, the EHVE would improve its power and efficiency by further 30% compared to the results presented. The EHVE design also has the advantage of less complicated technology at the manufacturing stage than that required for Stirling engines. The use of modern manufacturing solutions, as applied to typical internal combustion engines, would allow a construction of the EHVE.

One can speculate that EHVEs might be used on small nuclear powered submarines as they are as quiet as Stirling engines. Strong sunlight may deliver sufficient energy to run the EHVE. The power provided by this engine could be further increased through the use of a tracking device to maximise the solar energy harvested. Considering the above mentioned advantages, it can be expected that this new engine will raise interest among the world energy community. In other words, the proposed engine combines the best of two worlds – the well-known technology of manufacturing of internal combustion engines and an environment friendly external

burning process (among other available sources of heat) – and can stimulate a wide range of applications of externally heated engines in the near future at a reasonable cost.

A lack of any valves is an important advantage of the Stirling engine, but we need to emphasise the advantages of the EHVE:

- (1) the working gas is air, whereas the Stirling engines considered have to use helium to get a comparable level of power and efficiency;
- (2) the EHVE uses a less complicated, conventional crankshaft system, see Fig. 2, met in most modern internal combustion engines;
- (3) lack of a heat regenerator (additional heat exchanger), one of the essential parts of Stirling engines, in the EHVE makes the heat exchange system simpler;
- (4) another advantage of not using a regenerator (usually made of steel wires) is such that the EHVE can be lubricated with the same oil system as in internal combustion engines.

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